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ENERGY HARVESTING FROM TRANSVERSE GALLOPING USING A FLEXIBLE CRANK-ROD

Sorribes-Palmer, F., Sanz-Andres, A. and Alonso, G.

IDR/UPM, E.T.S.I. Aeronáuticos
Universidad Politécnica de Madrid
Pza. Cardenal Cisneros, 3
28040 Madrid, Spain
{felix.sorribes, angel.sanz.andres, gustavo.alonso}@upm.es

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Abstract. *In this paper a coupled fluid-structure model of a flexible crank-rod connected to an electrical generator is analyzed and compared with other ways to extract energy from transverse galloping [13]. One of the problems of energy harvesting is the conversion of linear oscillatory motion to a type of motion more compatible with an energy conversion device. One possibility is to transform the linear motion into a rotational one. The model considers a two degree-of-freedom galloping oscillator where fluid forces are described resorting to a quasi-steady condition by using a quasi-steady aerodynamics model. Transverse galloping is a dynamic instability in which IDR/UPM (Instituto de Microgravedad Ignacio Da Riva, Universidad Politécnica de Madrid) has a large interest [12, 3, 4]. This instability can be used for energy extraction. One possible application could be as a damping system for pressure waves in high speed railway tunnels. The wind speed induced by a pressure waves generated by a high speed train during its transit through a railway tunnel can surpass 5 m/s, and the train wake can induce speeds over 25 m/s. Due to the nature of the excitation and with a proper location for the device in the tunnel, these can be an optimal emplacement for devices based on aeroelastic instabilities such as transverse galloping.*

1 INTRODUCTION

The interest on energy harvesting in slow flows by aeroeslastic instabilities has recently increased [3, 12, 2, 5, 6, 8, 9, 11]. One of the key factors to extract energy from a variable amplitude oscillations, as it can be the case of transverse galloping, is the conversion of this kinetic energy into electrical energy which could be accumulated in batteries for later use. This mechanical energy can be converted by piezoelectric, electrostatic, or electromagnetic means.

Transverse galloping is caused by a coupling between the aerodynamic forces and the across-wind oscillations induced in the structure, this changes the angle of attack, which in turn varies the aerodynamic forces modifying the dynamical response of the structure. The structure, which usually has low stiffness and low damping, moves in the direction normal to the average wind speed. This motion is characterized by oscillations of large amplitude and low frequency. The most important parameters which influence galloping (considered here as a one-degree-of-freedom oscillator subjected to aerodynamic forces) are: the geometric shape, the angle of incidence, the free stream flow speed, the density and viscosity of the fluid, the turbulence intensity of the flow, and the system's mechanical properties (mass, stiffness and damping).

The optimization of energy extraction from transverse galloping using electromagnetic generator is studied in [13]. A electro-fluid-elastic model is introduced to describe the coupling between the galloping body and the linear motion electromagnetic generator. The maximum efficiency of the proposed mechanism is achieved when the speed of the flow is twice the speed to trigger transverse galloping. Also an expression for the optimal electrical load resistance as a function of the reduced velocity is presented.

In [14] a dual mass system as a method to enhance energy extraction from transverse galloping is discussed. Three different configurations are analyzed. The importance of parasitic damping to improve energy harvested in a broader band of incident flow velocities is shown.

Geometry restrictions or flow conditions (as gust flows, wakes of array devices, high turbulent flows, etc.) that could lead to aeroelastic problems on a wind turbine can be an opportunity for transverse galloping devices.

Another example are the ducted wind turbines with shrouded inlets used in energy harvesting at low flow speeds. Some are used for to deliver power to wireless sensors as shown in [7]. The concepts of wind turbines with double diffusers are studied in [10]. Turbines need a constant flow to be efficient and also a big air volume around them.

A possible location for transverse galloping devices could be in railway tunnels for high speed trains. The flow induced by the pressure waves generated during the passing train through the tunnel is enough to trigger galloping.

When a train approaches a tunnel entrance portal, the air inside the tunnel begins to be compressed due to the open-air local field preceding the train nose. Once the train head reaches the tunnel entrance, the pressure increases faster until the nose is completely inside the tunnel. This interaction between the tunnel entrance hood and the train nose determines the shape of the compression wave generated, but the total pressure gauge Δp_N remains constant. The pressure continues to increase slowly due to the viscous effects on the tunnel wall and the train surface Δp_{fr}, until the end of the train reaches the tunnel entrance portal, and a rarefaction wave is then generated with a pressure decrease Δp_T. The pressure decreases suddenly Δp_{HP} when the train nose passes the measurement point. These pressure variations are known as the train reference signature, and is defined in the standard [1] (see Figure 1).

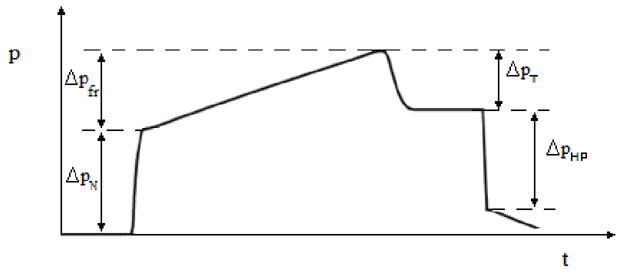

Figure 1: Train reference signature defined by the standard [1].

As a rough estimation of the amount of available energy inside a railway tunnel for transverse galloping devices, lets consider a train of 200 m length entering at 300 km/h. The characteristic time of the pressure wave generated is 2.4 s, the pressure gauge of the wave generated by a tunnel-train combination is around 1-3 kPa, this depends on the blockage and of the altitude of the tunnel entrance. Idealizing the pressure wave as a squared pulse of 2 kPa, the speed induced by the wave will be close to 5 m/s. The average energy density per unit area of a plane wave is 15 N/m^2. Assuming that each device has a frontal area of 0.2 m^2 and the efficiency of conversion is 10 % the extractable energy by each device would be 1.5 W. When using the flow speed induced by the wake of the train (speed can rise up to 20 m/s depending on train speed and blockage ratio) this extractable energy can rise till 100 W per galloping device. Assuming a logarithmic decrement of 0.01 with oscillation frequency of 2 Hz, the time to reduce the oscillation to a 10 % of the maximum amplitude would be approximately 10 seconds. In long enough tunnels, this time could be used to extract energy before the next reflection reaches the device again. In this moment the speed induced by the wave would be in the opposite direction. If the body shape used in the device had a symmetric behavior, the energy extracted or damped would be of the same order in both directions. A matrix configuration could be distributed around different sections in the tunnel and optimally separated with higher flow speeds. These devices could be also used as remote sensors for tunnel illumination.

In this paper a flexible crank-rod mechanism to transform the oscillations of variable amplitude of transverse galloping into a rotary motion of an electric generator is presented.

2 MATHEMATICAL MODEL

Let us consider a galloping body of mass M_1 attached to another mass M_2 through a spring of stiffness K. The mass M_2 is attached to a system crank-rod to an electric generator with moment of inertia I and a rotation angle θ. The scheme of the mechanism is shown in Figure 2.

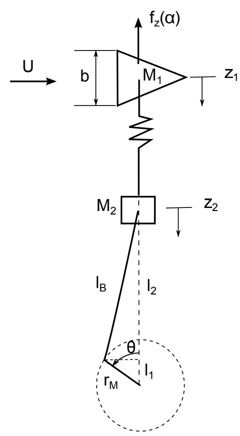

Figure 2: Scheme of a flexible crank-rod galloping mechanism.

The kinetic and potential energy of the mechanism are

$$
\begin{aligned}
T &= \frac{1}{2}M_1\dot{z}_1^2 + \frac{1}{2}M_2\dot{z}_2^2 + \quad \frac{1}{2}I\dot{\theta}^2 \\
U &= \frac{1}{2}K\left(z_2 - z_1\right)^2 \qquad ,
\end{aligned}
\tag{1}
$$

where z_1 denotes the position of the galloping body and z_2 the position of mass M_2. The parameters z_2 and θ are related through the expression

$$
\begin{aligned}
l_1 + l_2 = r_M\cos\theta + \sqrt{l_B^2 - r_M^2\sin^2\theta} &= r_M\cos\theta + l_B\sqrt{1 - \tfrac{r_M^2}{l_B^2}\sin^2\theta} \\
\simeq r_M\cos\theta + l_B\left(1 - \tfrac{1}{2}\tfrac{r_M^2}{l_B^2}\sin^2\theta\right) &= r_M\cos\theta + l_B\left(1 - \tfrac{1}{2}\tfrac{r_M^2}{l_B^2}\tfrac{1-\cos^2 2\theta}{2}\right) \\
&= r_M\cos\theta + l_B\left(1 - \tfrac{1}{4}\tfrac{r_M^2}{l_B^2} + \tfrac{1}{4}\tfrac{r_M^2}{l_B^2}\cos 2\theta\right) \\
&= r_M\cos\theta + \tfrac{r_M}{4}\left(\tfrac{r_M}{l_B}\right)\cos 2\theta - l_B\left(1 - \tfrac{1}{4}\tfrac{r_M^2}{l_B^2}\right) \ ,
\end{aligned}
\tag{2}
$$

considering $r_M \ll l_B$, so then the displacement of the mass M_2 is $z_2 \simeq -r_M\cos\theta$ and its speed $\dot{z}_2 = r_M\theta\sin\theta$. The kinetic energy can be written as

$$
T = \frac{1}{2}M_1\dot{z}_1^2 + \frac{1}{2}M_2 r_M^2\dot{\theta}^2\sin^2\theta + \frac{1}{2}I\dot{\theta}^2. \tag{3}
$$

Applying Lagrange equations

$$
M_1\ddot{z}_1 + K\left(z_1 - z_2\right) = f_z\left(\alpha\right) \tag{4}
$$

$$
M_2 r_M^2[\sin 2\theta\dot{\theta}^2 + \sin^2\theta\ddot{\theta}] + I\ddot{\theta} - M_2 r_M^2\dot{\theta}^2\sin\theta\cos\theta - K\left(z_1 - z_2\right) r_M\sin\theta = M(\dot{\theta}) \tag{5}
$$

where $f_z(\alpha)$ is the aerodynamic transverse force applied to the mass M_1 immersed in the incoming flow and $M(\dot{\theta})$ is the torque applied by the electric generator. The relationship $\frac{\mathrm{d}}{\mathrm{d}t}\left[\sin\theta\dot{\theta}\right] = 2\sin\theta\cos\theta\dot{\theta}^2 + \sin^2\theta\ddot{\theta}$ has been used.

2.1 Simplified case $M_2 = 0$

In the case that $M_2 = 0$ the Eqs. 4 and 5 can be simplified as follows

$$
M_1\ddot{z}_1 + K\left(z_1 - z_2\right) = f_z\left(\alpha\right) \tag{6}
$$

$$
I\ddot{\theta} - K\left(z_1 - z_2\right) r_M\sin\theta = M(\dot{\theta})\,. \tag{7}
$$

A quasi-steady solution can be obtained under the assumption that if the moment of inertia I is large enough then the speed of the electric generator is almost constant $\dot{\theta} \simeq \Omega_0$, and then $\theta = \Omega_0 t + \varepsilon\theta_1$. The equations can be rewritten

$$M_1 \ddot{z}_1 + K\left(z_1 + r_M \cos\theta\right) = f_z\left(\alpha\right) \tag{8}$$

$$I\varepsilon\ddot{\theta}_1 - K\left(z_1 + r_M \cos\Omega_0 t\right) r_M \sin\theta = M(\dot{\theta})\,, \tag{9}$$

if the mass M_1 is oscillating due to galloping at the resonance frequency of the system $\omega_0 = \sqrt{K/M_1}$ and the crankshaft of the electric generator turns at the same angular frequency $\Omega_0 = \omega_0$. Then Eq. 6 can be rewritten

$$M_1 \ddot{z}_1 + K z_1 = f_z\left(\alpha\right) - K r_M \cos\Omega_0 t = F\,, \tag{10}$$

which is the equation of the forced oscillator, under the action of a force F. The aerodynamic force in the direction transverse to the flow $f_z(\alpha)$, is

$$f_z(\alpha) = \frac{1}{2}\rho b U^2 C_z(\alpha)\,, \tag{11}$$

where $\tan\alpha = (1/U)\dfrac{\mathrm{d}z_1}{\mathrm{d}t}$, U the velocity of the incident flow, ρ the fluid density, b the characteristic dimension of the body in the direction transverse to the flow, and $C_z(\alpha)$ is the dimensionless coefficient of the instantaneous aerodynamic force on the galloping body in the transverse direction to the incident flow

$$C_z(\alpha) = -\frac{C_l(\alpha) + C_d \tan\alpha}{\cos\alpha} = \sum a_n \tan^n \alpha\,. \tag{12}$$

This expression refers to the study of the transverse oscillatory motion at a given pitch angle of the body θ_p considered as reference, with $\theta_p = 0$ as origin. If the lift $c_l(\theta_p)$ and drag $c_d(\theta_p)$ coefficients determined in static conditions ($\theta_p = \alpha$) are defined as a function of the pitch angle of the body θ_p, and the reference point of study is $\theta_{p0} \neq 0$, doing the change of variable $\alpha = \theta_p - \theta_{p0}$, the functions $c_l(\alpha)$ and $c_d(\alpha)$ can be generated for each θ_{p0}. The quasi-steady hypothesis is assumed, in which the timescale of the oscillating body is much larger than the residence timescale of the flow particles. The critical speed for transverse galloping is

$$U_{cr} = 4m\omega\zeta/(\rho b H)\,, \tag{13}$$

where $H = \dfrac{\partial c_l}{\partial\alpha} + c_d$ and ζ is the damping coefficient. In this model, the damping or energy extraction is produced by the torque of the electric generator.

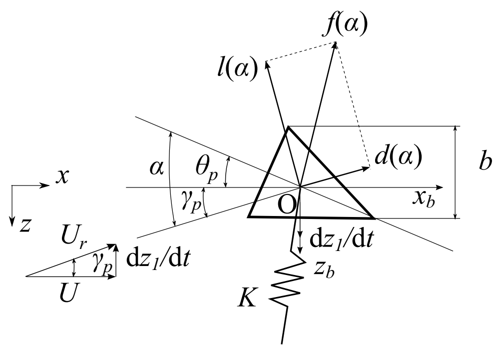

Figure 3: Schematic representation of aerodynamic forces and the angle of attack, α, pitch angle of the velocity, γ_p, pitch angle of the body, θ_p; z, x is the inertial reference frame and z_b, x_b is the reference frame attached to the body.

There must exist a phase delay φ between the displacement z_1 and the position angle of the crank-rod, for the transmitted power by the spring averaged in a cycle not being zero. So then

$$z_1 = A\cos(\omega_0 t - \varphi) \qquad \dot{z}_1 = -A\omega_0 \sin(\omega_0 t - \varphi)\,. \tag{14}$$

The extracted power averaged per cycle is

$$P_e = \frac{1}{T}\int_0^T f_z \frac{\mathrm{d}z_1}{\mathrm{d}t}\mathrm{d}t = \frac{\rho U^2 b}{2T}\int_0^T C_z \frac{\mathrm{d}z_1}{\mathrm{d}t}\mathrm{d}t\,, \tag{15}$$

and the transmitted power averaged per cycle is

$$\begin{aligned} P_t &= \tfrac{1}{T}\int_0^T \left(-K r_M \cos\omega_0 t\right)\tfrac{\mathrm{d}z_1}{\mathrm{d}t}\mathrm{d}t = \tfrac{1}{T} K r_M \omega_0 A \int_0^T \cos(\omega_0 t)\sin(\omega_0 t - \varphi)\mathrm{d}t \\ &= -\frac{1}{2} K r_M \omega_0 A \sin\varphi\,. \end{aligned} \tag{16}$$

The equilibrium of the oscillator of Eq. 10 is reached when $P_e + P_t = 0$. Analyzing the rotational motion, integrating Eq. 9 multiplied by $\dot{\theta}$

$$\varepsilon I \int_0^T \ddot{\theta}_1 \dot{\theta}\mathrm{d}t + K\int_0^T \left(z_1 + r_M \cos\omega_0 t\right) r_M \sin\omega_0 \dot{\theta}\mathrm{d}t = \int_0^T M(\omega_0)\dot{\theta}\mathrm{d}t \tag{17}$$

as we are assuming periodic motion, the first part of the integral is zero. And as $\dot{\theta} \simeq \omega_0$

$$K\int_0^T \left[\left(A\cos(\omega_0 t - \varphi) r_M \sin\omega_0 + r_M^2 \sin\omega_0 \cos\omega_0 t\right)\omega_0\right]\mathrm{d}t = \int_0^T M(\omega_0)\omega_0 \mathrm{d}t \equiv P_t T\,. \tag{18}$$

The first term is the transmitted power

$$P_t T = -\frac{T}{2} K r_M \omega_0 A \sin\varphi\,. \tag{19}$$

The dimensionless extracted power averaged per cycle is

$$\begin{aligned} p_e &= \frac{P_e}{\frac{1}{2}\rho b U^3} = \tfrac{1}{UT}\int_0^T C_z \tfrac{\mathrm{d}z_1}{\mathrm{d}t}\mathrm{d}t = \tfrac{1}{UT}\int_0^T \sum a_n \tan^n(\alpha)\tfrac{\mathrm{d}z_1}{\mathrm{d}t}\mathrm{d}t \\ &= \frac{1}{2}\left[a_1\left(\frac{A^*}{U^*}\right)^2 + \frac{3}{4}a_3\left(\frac{A^*}{U^*}\right)^4 + ...\right]. \end{aligned} \tag{20}$$

A typical dimensionless extracted power curve, as a function of the amplitude A^*/U^* is shown in Figure 2.1. There is a maximum extracted power point p_{emax} which is attained at $(A^*/U^*)_{max}$. The dimensionless transmitted power is defined as

$$p_t = \frac{P_t}{\frac{1}{2}\rho b U^3} = \frac{A\Omega_0 K r_M \sin\varphi}{\frac{1}{2}\rho b U^3} = \frac{A\Omega_0^3 M_1 r_M \sin\varphi}{\frac{1}{2}\rho b U^3} = \frac{2m^* \sin\varphi}{U^*}\left(\frac{A^*}{U^*}\right)^2 \frac{r_M^*}{A^*}\,, \tag{21}$$

where the amplitude of the oscillations has been divided by the characteristic dimension of the body in the direction transverse to the flow b, $A^* = \dfrac{A}{b}$, $r_M^* = r_M/b$, the mass of the body by the characteristic mass of the surrounding fluid, $m^* = M_1/(\rho b^2)$; the spring stiffness is $K = \Omega_0^2 M_1$ and the dimensionless velocity $U^* = U/(\omega_0 b)$. Comparing the Eq. 21 with the classic of the dissipated power [12]

$$p_d = \frac{2m^*\xi}{U^*}\left(\frac{A^*}{U^*}\right)^2 , \tag{22}$$

the $\sin\varphi$ is taking the role of the damping coefficient ξ, and r_M^*/A^* is replacing A^{*2}.

On one side the expression for the dimensionless dissipated power in the electric generator is

$$p_d = \frac{1}{T}\frac{1}{\frac{1}{2}\rho b U^3}\int_0^T M(\omega)\omega \mathrm{d}t = \frac{M(\omega_0)\omega_0}{\frac{1}{2}\rho b U^3} , \tag{23}$$

where it has been assumed that $\omega = \omega_0$ is a constant. Under the assumption of a electric generator with linear behavior, $M(\omega_0) = k_M\omega_0$ from Eq. 23 is

$$p_d = \frac{k_M\omega_0^2}{\frac{1}{2}\rho b U^3} = \frac{k_M}{\frac{1}{2}\rho b^4\omega_0}\frac{1}{U^{*3}} = \frac{C_M}{U^{*3}} , \tag{24}$$

where $C_M = 2k_M/(\omega_0\rho b^4)$. The balance of power transmitted Eq. 16 and dissipated Eq. 24 at the electric generator leads to

$$\frac{1}{2}M_1\omega_0 A r_M \sin\varphi \quad = k_M . \tag{25}$$

From this equation the relation between φ and A is obtained

$$\sin\varphi = \frac{2k_M}{M_1\omega_0 A r_M} \tag{26}$$

On the other hand, imposing that $p_e = p_d$ from Eq. 20 and Eq. 24

$$p_e = \frac{1}{2}\left[a_1\left(\frac{A^*}{U^*}\right)^3 + \frac{3}{4}a_3\left(\frac{A^*}{U^*}\right)^4 + ...\right] = \frac{C_M}{U^{*3}} = p_d , \tag{27}$$

so p_d can be expressed as a function of A^*/U^*

$$p_d = \frac{C_M}{A^{*3}}\left(\frac{A^*}{U^*}\right)^3 . \tag{28}$$

In this way it is easy to find the condition for maximum extractable power (see Figure 2.1). The process to obtain the maximum power is the following, as the extractable power can be calculated directly from the test data as a function of A^*/U^* and at the same time it is $p_e = p_{emax} = C_M/U^{*3}$, the dimensionless speed can be calculated $U^{*3} = p_{emax}/C_M$. As $(A^*/U^*)_{max}$is known by using $A^*/U^* = (A^*/U^*)_{max}$, hence

$$A^* = U^*\left(\frac{A^*}{U^*}\right)_{max} = \sqrt[3]{\frac{p_{emax}}{C_M}}\left(\frac{A^*}{U^*}\right)_{max} . \tag{29}$$

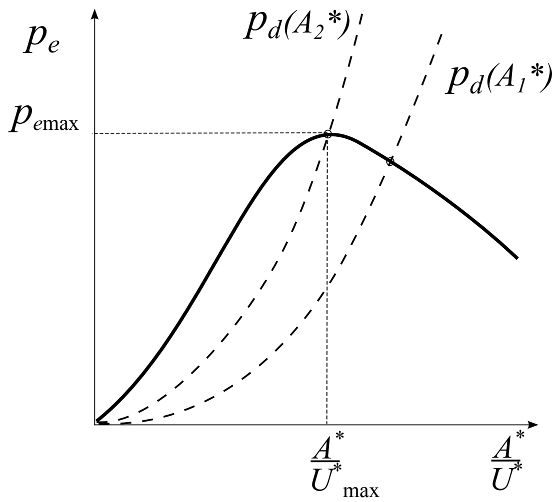

Figure 4: Determination of the maximum dimensionless extracted power p_{emax}. Variation of dimensionless extracted power, p_e , as a function of relative amplitude, $\frac{A^*}{U^*}$, for two values of the dimensionless amplitude oscillations A^*.

3 EXTRACTABLE ENERGY FROM DIFFERENT GALLOPING BODIES

From the experimental data in [12] where different bodies shapes were tested in the wind tunnel we obtain $f_z(\alpha)$, and using a characteristic $M(\omega)$ of a common electric electric generator, we can calculate the extractable energy from the flow by using a device based on transverse galloping and using a linear to rotatory energy conversion mechanism. The parameters used for the evaluation of the relative amplitude of the oscillations summarized in Table 1 have been selected based on [13].

Characteristic transverse dimension of the body, b [m]	0.1
Mass per unit length, m [kg/m]	0.612
Crank-shaft dimension, r_M [m]	0.05
Density of the fluid, ρ [kg/m^3]	1.225

Table 1: Values of the parameters used to estimate the dimensionless dissipated power, p_d.

In Table 2 is presented the maximum extractable power per averaged per cycle, made dimensionless by using the maximum relative amplitude $A^*/U^*|_{p_{emax}}$ in a cycle (or specific power) for different galloping body shapes,

$$\tilde{p}_{emax} = \frac{p_{emax}}{\left.\frac{A^*}{U^*}\right|_{p_{emax}}} , \tag{30}$$

and the pitch angle of the body at which this energy is extracted.

In Figure 5 is shown the specific extracted power $\tilde{p}_{\mathrm{emax}}$ for the body shapes indicated in Table 2, and the specific dissipated power $\tilde{p}_d$

$$\tilde{p}_d = \frac{C_M}{A^{*3}} \left(\frac{A^*}{U^*}\right)^2 . \tag{31}$$

Body shape	$\frac{A^*}{U^*}\Big\vert_{\tilde{p}_{emax}}$	$\tilde{p}_{emax}$	θ_p [rad]
D-shape	0.33	0.19	1.42
Triangle $\beta = 30^{\rm o}$	0.33	0.11	3.16
Triangle $\beta = 40^{\rm o}$	0.47	0.13	3.18
Triangle $\beta = 60^{\rm o}$	0.77	0.14	3.14

Table 2: Efficiency of galloping body to convert energy through linear to rotatory mechanism.

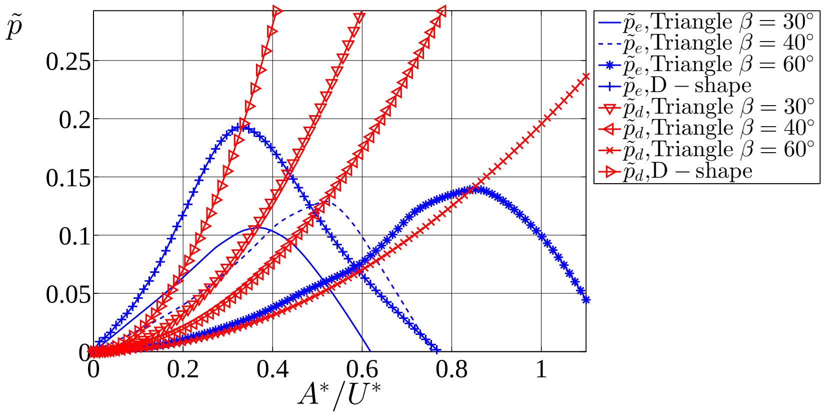

Figure 5: Determination of the maximum dimensionless extracted power $\tilde{p}_{\rm emax}$. Variation of dimensionless extracted power, $\tilde{p}_e$, as a function of relative amplitude, $\frac{A^*}{U^*}$.

The stiffness of the spring would be selected depending on the body shape chosen. The phase delay can be modified as the flow speed conditions vary by modifying the torque of the electric generator (a multiplication system) in order assure maximum power extraction. In Table 3 are indicated the reduced amplitude of the oscillations, reduced speed, the phase delay, the angular frequencies for the stiffness of the different strings selected and with $k_M = 10^{-3}$.

Body shape	K [N/m]	C_M	A^*	U^*	φ [rad]	ω [rad/s]
Triangle $\beta = 30^{\rm o}$	10	4.04	1.59	4.89	0.076	4.04
Triangle $\beta = 40^{\rm o}$	95	1.31	1.3	2.79	0.051	12.45
Triangle $\beta = 60^{\rm o}$	50	1.81	1.97	2.56	0.033	9.04
D-shape	60	1.65	0.98	2.87	0.067	9.90

Table 3: Mechanism parameters selected for transverse galloping energy harvesting.

In case of selecting $k_M = 10^{-2}\mathrm{Nm/(rad/s)}$ and $k = 6$ N/m and keeping the rest of the parameters the angular frequency will be $\omega = 3.13$ [rad/s]. The relative amplitude and reduced speed of oscillations at the equilibrium are presented in Table 4.

Body shape	A^*	U^*	φ [rad]
Triangle $\beta = 30^o$	3.74	11.46	0.58
Triangle $\beta = 40^o$	4.45	9.53	0.49
Triangle $\beta = 60^o$	6.04	7.87	0.35
D-shape	3.07	9.38	0.75

Table 4: Relative amplitude A^*, reduced speed $U^*$and phase delay φ of oscillations at the equilibrium.

A gearbox (a speed multiplication of approximately 20) should be used to make the electric generator work in an efficient point. In order to compare this results with other devices the mechanical efficiency of the mechanism η_M, and the efficiency of the electric generator η_E, must be taken into account. Assuming $\eta_M = 0.9$ and $\eta_E = 0.9$ the electric power extracted per unit area would be close to 0.16 mW/cm^{2}which is close to results presented in [7].

4 CONCLUSIONS

- The extractable power from oscillations of transverse galloping by using a linear to rotary conversion mechanism has been analyzed.
- The maximum extractable energy by using transverse galloping for different body shapes is presented.
- A method to design the energy harvesting mechanism, using an expression for the relative oscillation amplitude that maximizes the efficiency, is proposed.

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